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CON-ROD ACCELERATIONS, FORCES, DISPLACEMENTS

ADVANCED ENGINE DESIGN

SUPERVISION

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MEng Motorcycle Engineering

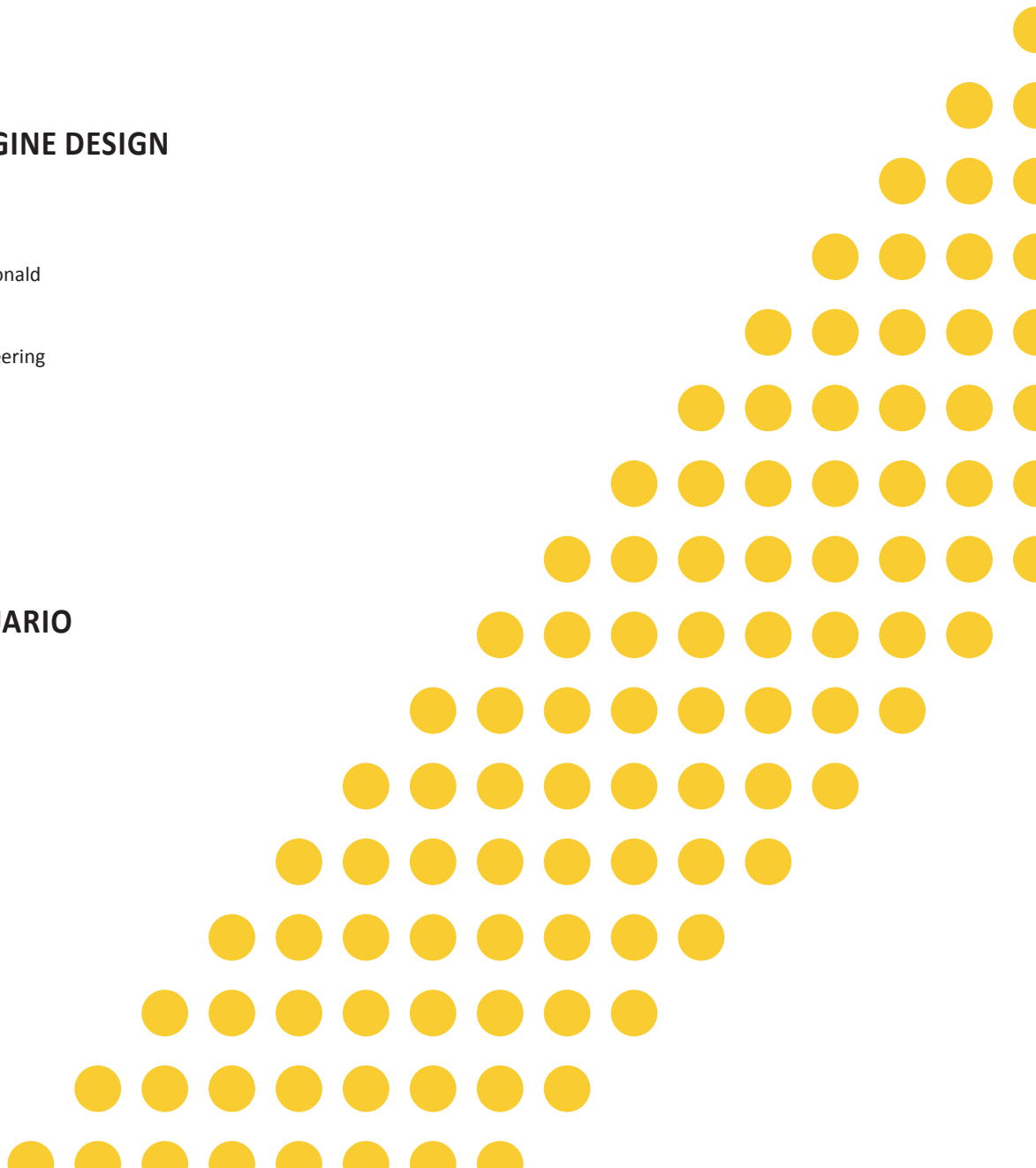
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1 INTRODUCTION

The following assessment was developed for the module of Advanced Engine Design of the third year – level 6 of the Motorcycle Engineering Degree.

The aim of the presented work is to develop simple analytical techniques to allow the design of the major structural components of an engine, throughout several theoretical and practical methods lectured in classes and sourced from other authors.

With cylinder pressure data from an engine provided and other data measured by the author, a simulation model will be developed to evaluate the con-rod's performance and the accuracy of the analytical techniques used.

Success will be measured by how accurate the results achieved are, allowing to relate con-rod accelerations, forces and displacements and the generalisation of this analysis for different engine configurations.

2 PRELIMINARY WORK

In the first chapter the engine pressure data and specifications, assumptions, as well as some other specific calculations that will be used throughout this report, will be defined.

This work was based on a Ford Sigma engine, using pressure data provided. See below a plot with the pressure data for all the given RPM values:

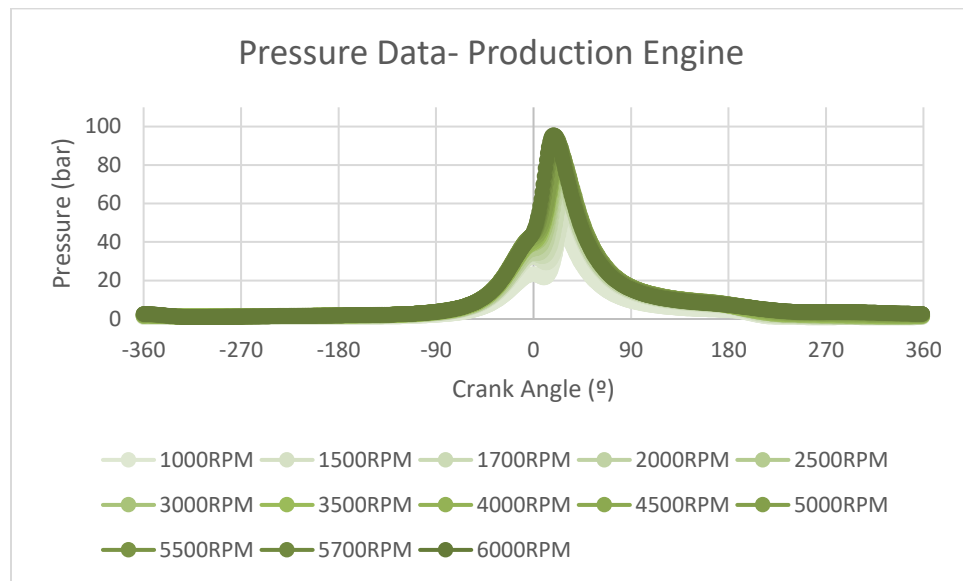


Figure 1. Ford pressure data per RPM over a 4-stroke cycle

The engine specifications required for the calculations developed in the next chapters are the following: (some of these values will be discussed in detail when needed in future calculations)

Engine Specifications		Units
Displacement	1596	cc
Nº of cylinders	4	
Bore	79	mm
Stroke	81.4	mm
Crank radius	40.7	mm
Conrod		Units
Conrod length	134	mm
Conrod/Crank ratio	0.30	
Conrod minimum	124.1	mm ²
Powder Forged UTS	750	N/mm ²
Powder Forged E	170000	N/mm ²
Powder Forget K	157440	N/mm
Titanium UTS	1100	N/mm ²
Titanium E	130000	N/mm ²
Titanium K	120396	N/mm

Masses		Units
Conrod	267.4	grams
Conrod cap	103.8	grams
Conrod bolts	35.19	grams
Conrod assembly	406.39	grams
Gudgeon pin	80.22	grams
Piston	257.7	grams
Piston ring	5.49	grams
Circlip	0.77	grams
Piston assembly	344.18	grams
Moving assembly		Units
Mass	0.480	Kg
Ring height	1.5	mm
Friction coeff	0.1	

Figure 2. Ford Sigma specifications

Because the presented work aims to evaluate an engine's components performance, it was desired to compare the production engine with a modified one - the latter one with more performance, increasing the stress in the engine.

The modified engine is capable of more RPM and the rate of increasing pressure was copied from previous RPM values. In a real situation, this increase in engine power could be obtained by using a different camshaft geometry, lighter parts, materials with less friction, etc.

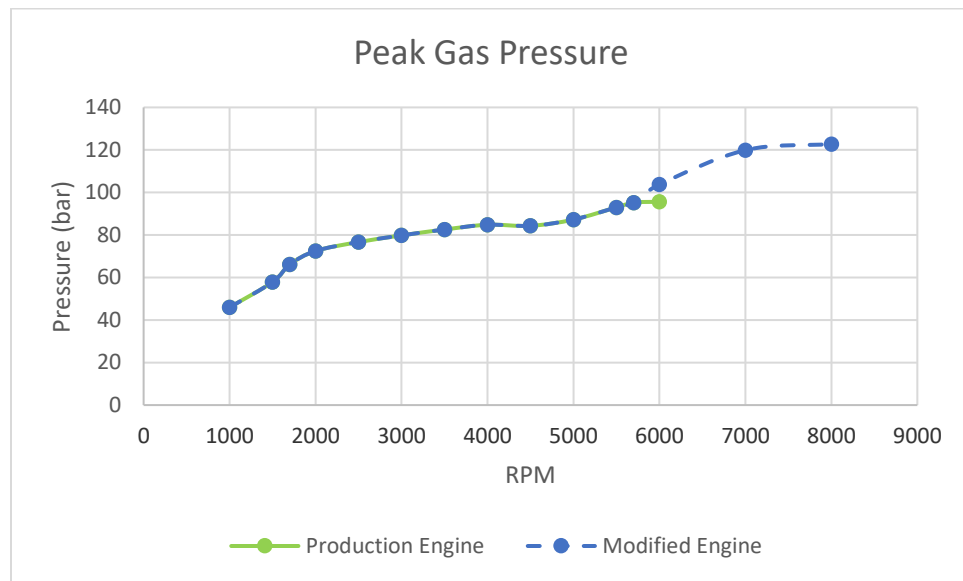


Figure 3. Peak Gas Pressure – Production Vs Modified Engine

3 PEAK GAS FORCES

3.1 Methodology

$$\mathbf{F_{Gas} = P_{Gas} * A}$$

Where,

- P_{Gas} is the average pressure of the 4 cylinders for each specific degree of crank angle in a certain RPM;

$$P_{Gas} = P_{from\ data} (bar) = P_{from\ data} * 10^5 (Pa)$$

- A is the piston area;

$$A = \frac{Bore^2 * \pi}{4} (m^2)$$

- F_{Gas} is the vertical force acting in the piston surface area (N);

3.2 Results

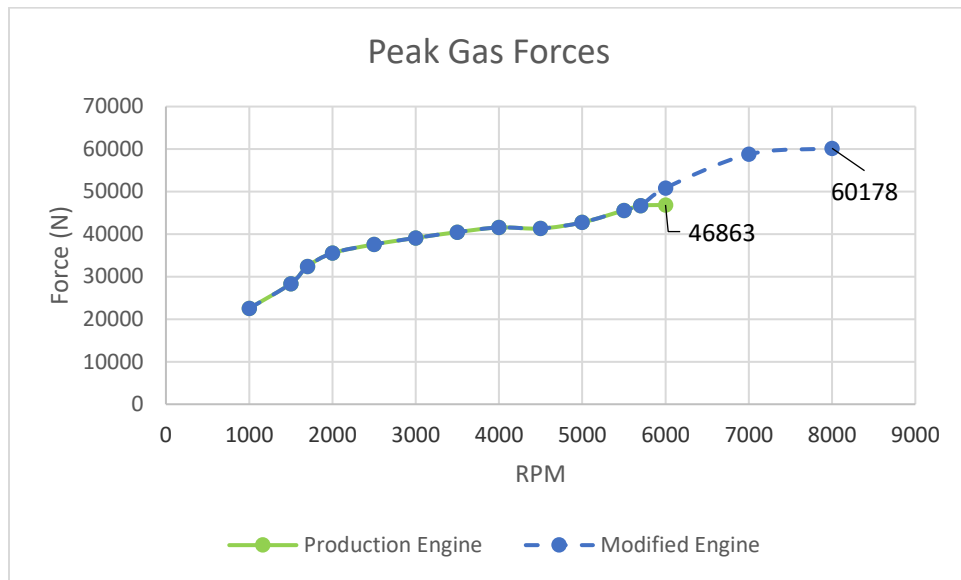


Figure 4. Peak Gas Forces - Production Vs Modified Engine

The first thing noticed about the peak gas forces plot, for each RPM, was that it presents the same profile as the peak gas pressures – due to the piston area being a constant.

- Peak Gas Force for Production Engine = 46863 N;
- Peak Gas Force for Modified Engine = 60178 N;

It can be easily concluded that, for the same piston area, bigger cylinder pressures result in a proportional increase of gas forces.

Note: A table with the values of Peak Gas Forces for each RPM is attached in the Appendix [Tab.1].

4 INERTIA LOADS ON ROD

4.1 Methodology

$$F_{Inertia\ on\ rod} = \frac{M_{Piston} * acc_{Piston}}{\cos \emptyset}$$

Where,

- $F_{Inertia\ on\ rod}$ is the inertia load acting in the direction of the rod (N);
- M_{Piston} is the moving mass [Figure 2];

$$M_{Piston} = M_{Piston\ assembly} + \frac{1}{3}M_{Conrod\ assembly} (kg)$$

- acc_{Piston} is the piston/moving assembly vertical acceleration (derivation in Appendix, Figure 27);

$$\frac{d^2S}{dt^2} = R * \omega * \cos \theta + \frac{R}{L} * R * \omega^2 * \cos 2\theta$$

- S = piston lift (m);
 - R = crank radius (m);
 - L = conrod length (m);
 - ω = angular velocity = $2\pi \frac{RPM}{60}$ (rad/s);
 - θ = crank angle (°);
- $\emptyset$ is the conrod angle (°) (derivation in Appendix, Figure 27);

4.2 Results

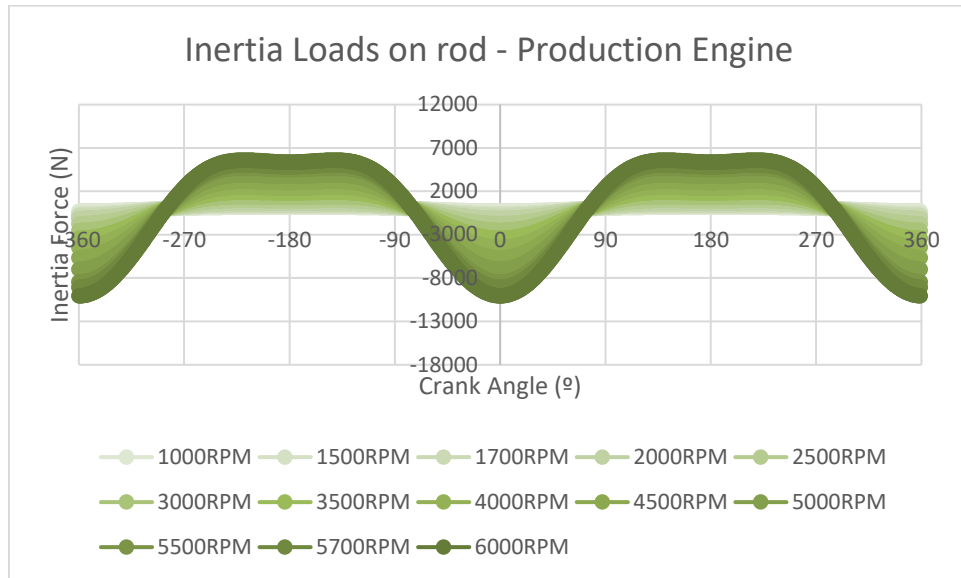


Figure 5. Inertia Loads on rod - Production Engine

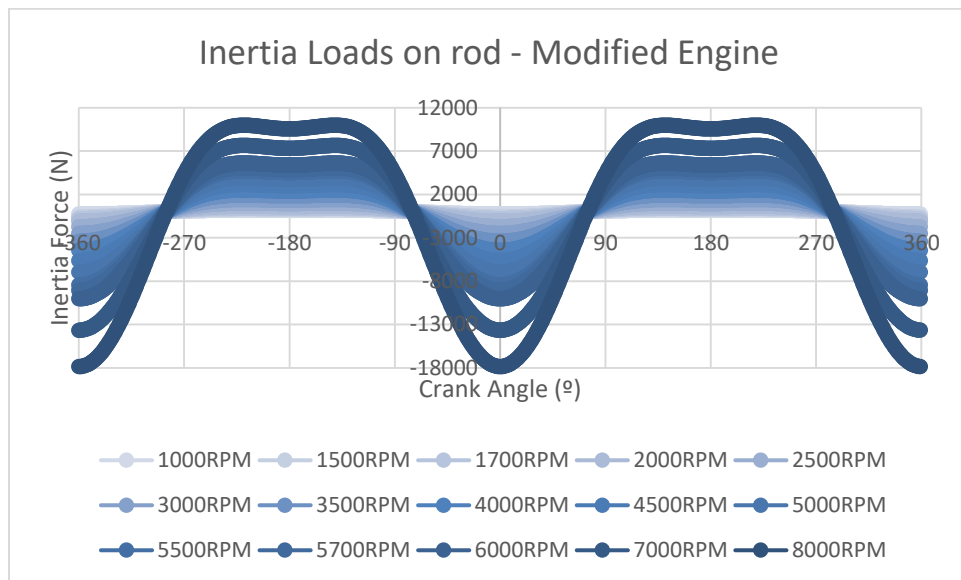


Figure 6. Inertia Loads on rod - Modified Engine

Due to the convection of signs chosen by the author, the results appear to be opposite to what was expect. It was decided that positive load values were in the direction of the con-rod (compression) and negative values were in opposite direction (tensile).

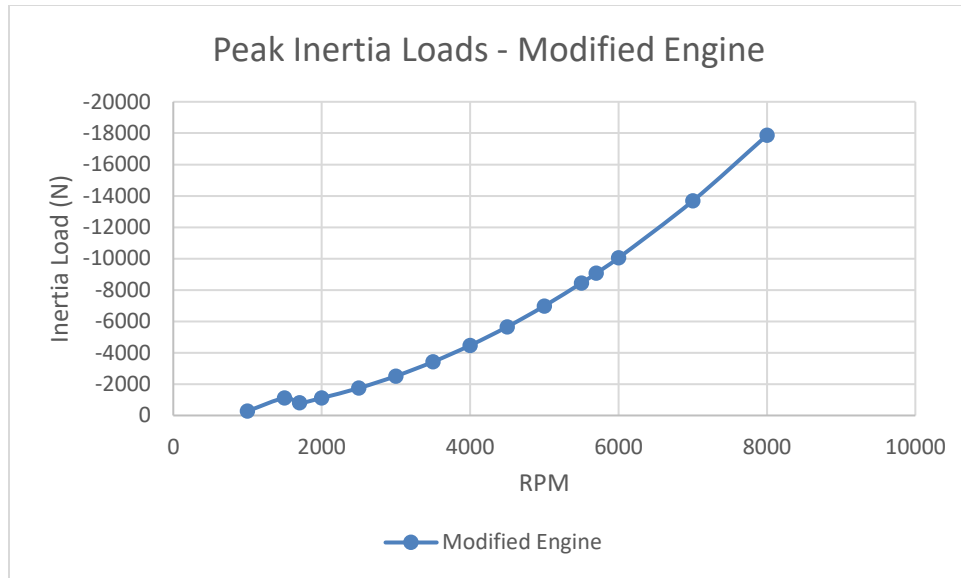


Figure 7. Peak Inertia Load Vs RPM

It is observed, as it was expected, that the inertia loads increase with higher RPM values, being this the only variable when the engine specifications are already defined. An increase of 77.8% of inertia loads in the con-rod, in the tensile direction, were calculated from increasing the RPM of the Modified Engine from 6000 to 8000.

- Peak Inertia Load for Production Engine@6000RPM = - 10048 N;
- Peak Inertia Load for Modified Engine@8000RPM = -17862 N;

Note: A table with the values of Peak Inertia Loads for each RPM is attached in the Appendix [Tab.1].

4.2.1 Inertia Loads Exercise

4.2.1.1 Single Plane Crank

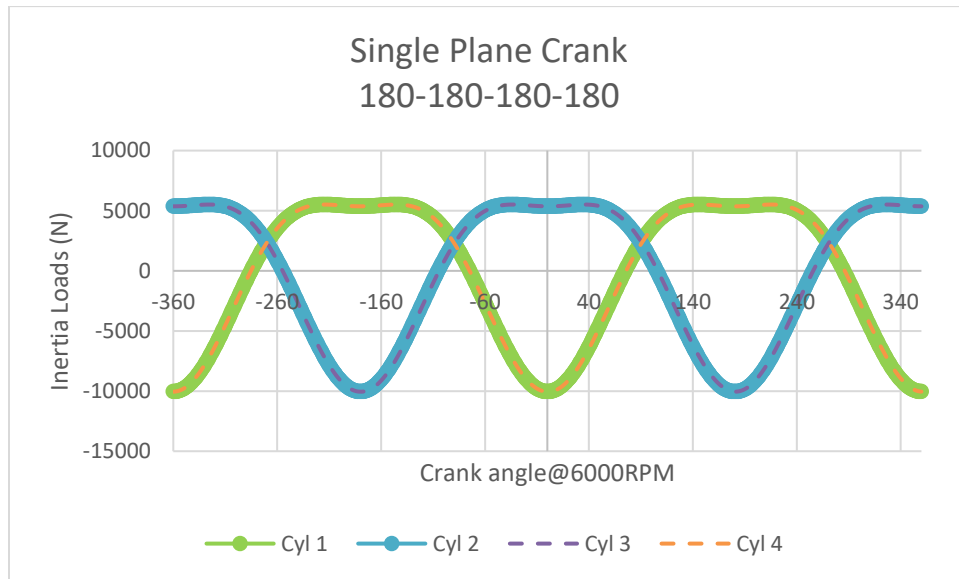


Figure 8. Inertia Loads in a Single Plane Crank with 4 cyl

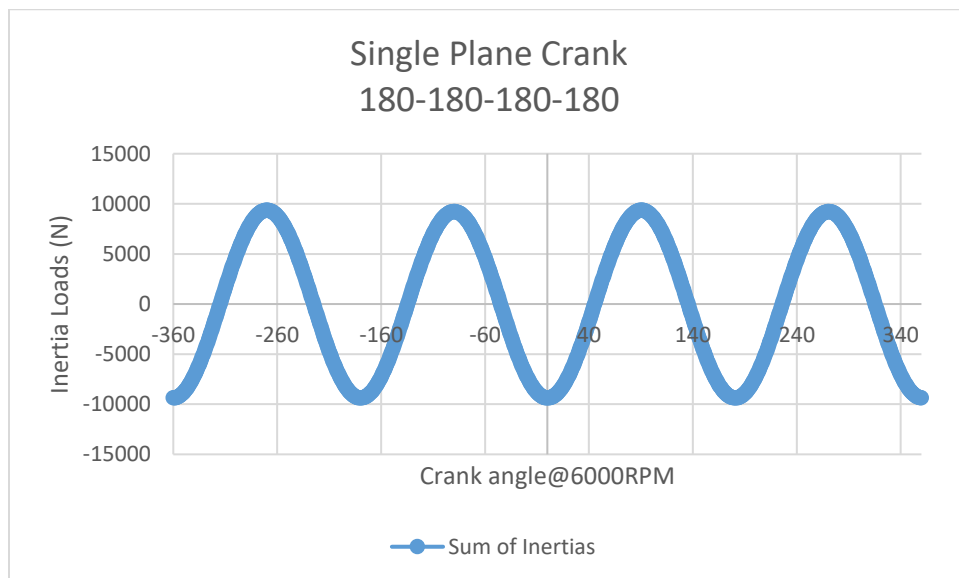


Figure 9. Sum of Inertia Loads in a Single Plane Crank with 4 cyl

With a Single Plane Crank and 4 cylinders, as the production engine being studied in this report, the sum of the Inertia Loads in the engine shows the following results:

- Same value for highest tensile inertia load;
- The compressive inertia load increased for the same value of the tensile inertia load;
- The inertia loads delivered during a 4-stroke cycle are very regular, symmetric in intensity and present 4 peak compressive and tensile loads per cycle;

4.2.1.2 Crossplane Crank

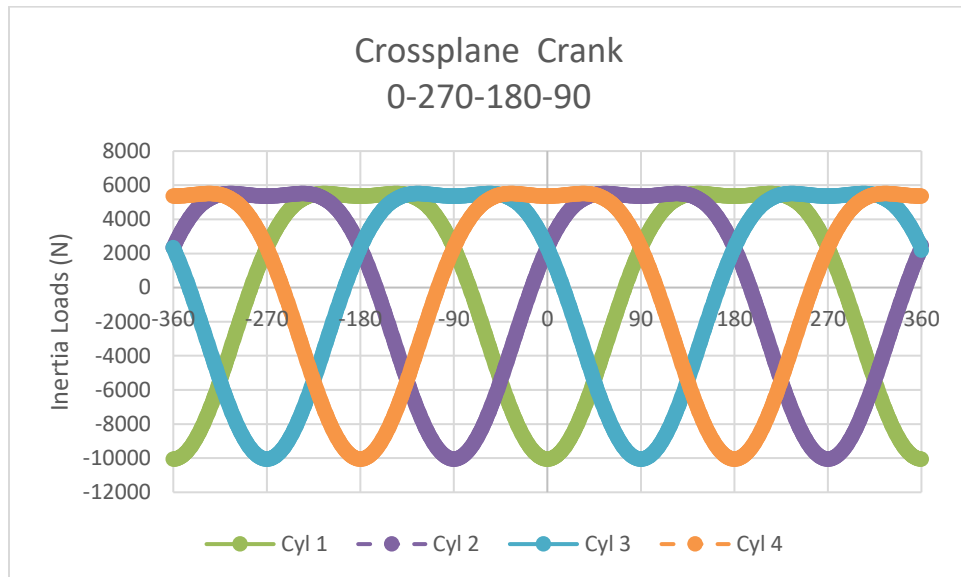


Figure 10. Inertia Loads in a Crossplane Crank with 4 cyl

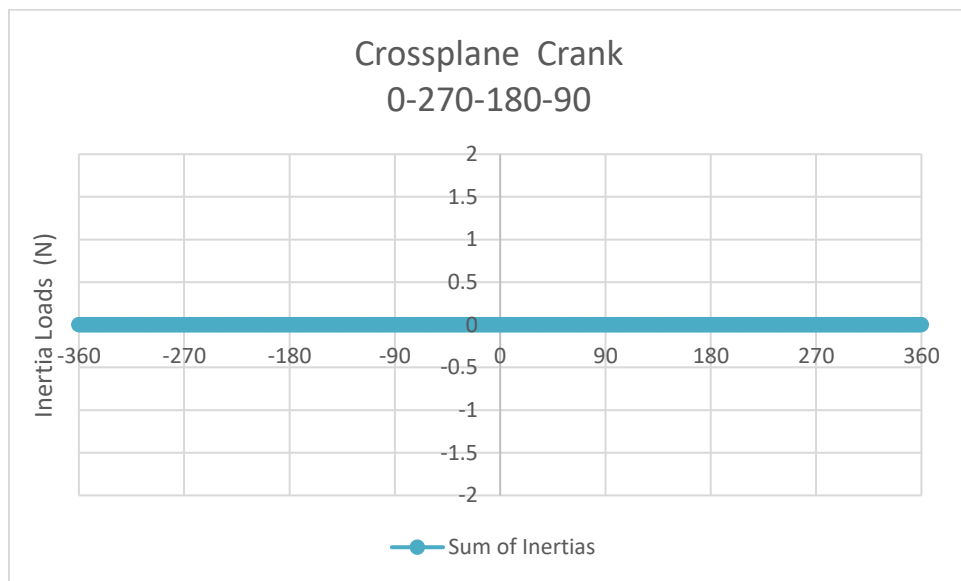


Figure 11. Sum of Inertia Loads in a Crossplane Crank with 4 cyl

Really interesting results were obtained when performing the sum of all inertia loads suffered by the pistons in a Crossplane geometry, during a 4-stroke cycle.

- The sum of the inertia loads from the 4 cylinders during a 4-stroke cycle is approximately zero!
- Due to this result, the Inertia Torque produced by the engine will be reduced substantially! (inertia Torque results in Chapter 12.5)

5 HIGHEST NET COMPRESSIVE FORCE ON ROD

5.1 Methodology

$$F_{Net\ on\ rod} = \frac{F_{Gas} + F_{Inertia}}{\cos \emptyset}$$

Where,

- $F_{Net\ on\ rod}$ is the net force in the rod – positive values mean the conrod is under compression forces, negative values mean the conrod is subjected to tensile loads (N);
- F_{Gas} is the vertical force acting in the piston surface area (N) [Chapter 3];
- $F_{Inertia}$ is the vertical inertia load acting in the moving mass (N) [Chapter 4];
- $\emptyset$ is the conrod angle (derivation in Appendix, Figure 27);

5.2 Results

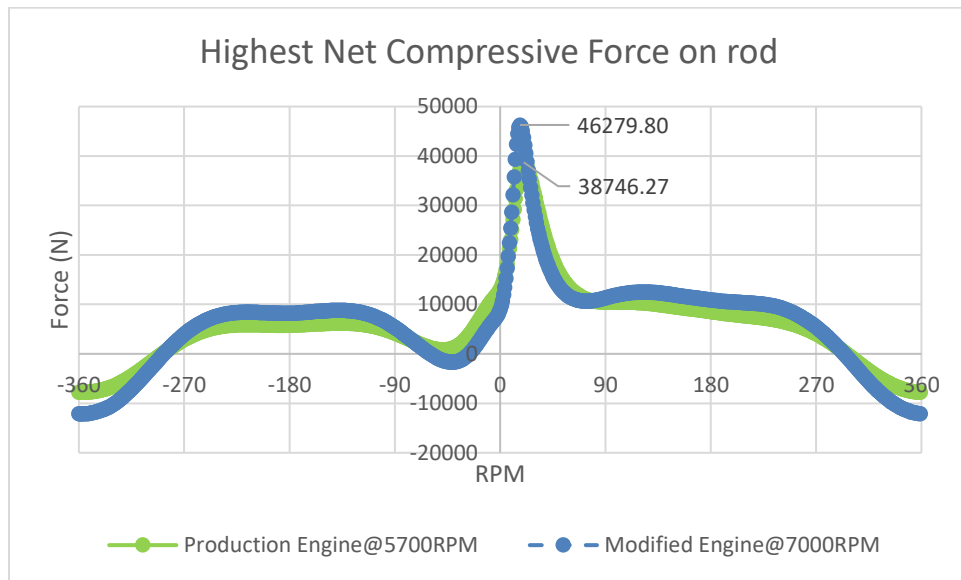


Figure 12. Net compressive force on rod

Interestingly the Highest Net Compressive Force, for the engines analysed, happens just before the maximum RPM available. Even though the Peak Gas Forces occur at maximum RPM [Chapter 3], the Peak Inertia Loads also occur at maximum RPM - having a negative contribution [Chapter 4].

- Highest Net Compressive Force for Production Engine = 38746 N;
- Highest Net Compressive Force Modified Engine = 46280 N;

Note: A table with the values of Highest Net Compressive Force for each RPM is attached in the Appendix [Tab.1].

6 HIGHEST NET TENSILE LOAD ON ROD

6.1 Methodology

$$F_{Net\ on\ rod} = \frac{F_{Gas} + F_{Inertia}}{\cos \emptyset}$$

Where,

- $F_{Net\ on\ rod}$ is the net load in the rod – positive values mean the rod is under compression load, negative values mean the rod is subjected to tensile loads (N);
- F_{Gas} is the vertical force acting in the piston surface area (N) [Chapter 3];
- $F_{Inertia}$ is the vertical inertia load acting in the moving mass (N) [Chapter 4];
- $\emptyset$ is the conrod angle (derivation in Appendix, Figure 27);

6.2 Results

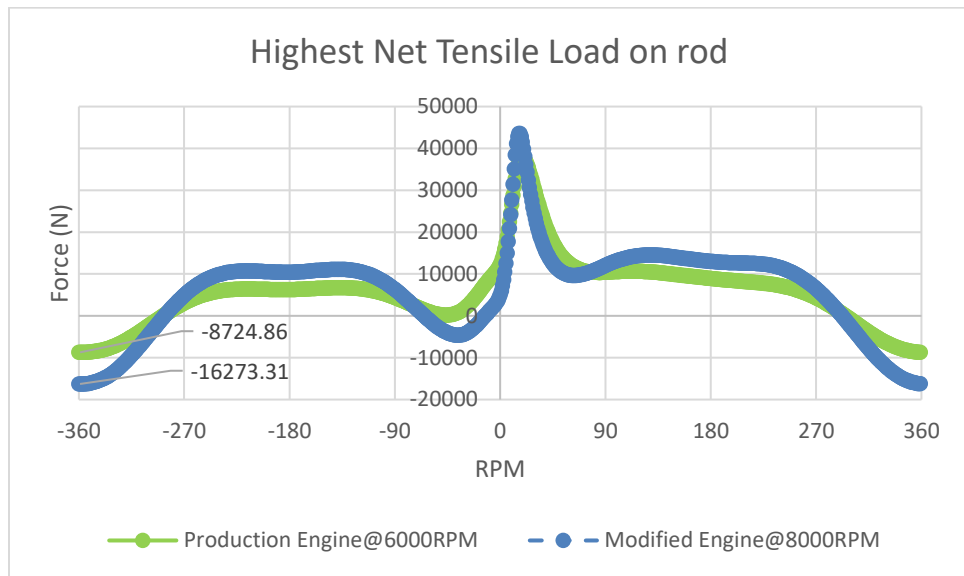


Figure 13. Net Tensile Load on rod

It was expected that the Highest Tensile Load, for the engines analysed, would take place at the maximum RPM available, since Peak Inertia Loads also occur at maximum RPM - having a negative contribution [Chapter 4].

- Highest Net Tensile Load for Production Engine = -8725 N;
- Highest Net Tensile Load for Modified Engine = -16273 N;

Note: A table with the values of Highest Net Tensile Load for each RPM is attached in the Appendix [Tab.1].

7 MAXIMUM COMPRESSION AND TENSILE STRESSES IN ROD SHANK

7.1 Methodology

$$\sigma_{compressive | tensile}(MAX) = \frac{F_{Net\ on\ rod\ (MAX)}}{A_{rod\ cross-sectional}}$$

Where,

- $\sigma_{compressive/tensile}$ is the maximum stress in the rod shank – positive values mean the conrod is under compression stress, negative values mean the conrod is subjected to tensile stresses (N/mm^2);
- $F_{Net\ on\ rod}$ is the net load in the conrod – positive values mean the conrod is under compression load, negative values mean the conrod is subjected to tensile loads (N) [Chapter 5 & 6];
- $A_{rod\ cross-sectional}$ is the rod shank minimum cross-sectional area (mm^2) (measurement and calculation in Appendix, Figure 28);

7.2 Results

	Rod shank cross-sectional area	Maximum Compression Force	Maximum Tensile Load	Maximum Compression stresses	Maximum Tensile Stresses
Production Engine	124.1	38746	-8725	312.2	-70.3
Modified Engine	124.1	46280	-16273	372.9	-131.1
Δ % of increase		19%	87%	19%	87%

Figure 14. Maximum Compression and Tensile Stresses in rod shank

It can be concluded that Stress is proportional to the Net Force applied on the rod and with the increasing performance, the Maximum Compression Stresses increase 19% and the Maximum Tensile Stresses are even more affected, increasing by 87%.

Note: A table with the values of Maximum Compression and Tensile stresses in the rod shank for each RPM is attached in the Appendix [Tab.1].

8 NOMINAL STATIC SAFETY FACTOR

8.1 Methodology

$$\text{SAFETY FACTOR} = \frac{UTS}{\sigma_{compressive | tensile}(MAX)}$$

Where,

- UTS is the ultimate tensile strength of the rod material, represents the maximum stress a material can withstand while being stretch before breaking (N/mm²);
- $\sigma_{compressive/tensile}$ is the maximum stress in the rod shank – positive values mean the conrod is under compression stress, negative values mean the conrod is subjected to tensile stresses (N/mm²) [Chapter 7];

8.2 Results

	Powder Forged Steel UTS	Maximum Compression stresses	Maximum Tensile Stresses	Safety Factor - Compressive Loads	Safety Factor - Tensile Loads
Production Engine	750	312	-70	2.4	10.7
Modified Engine	750	373	-131	2.0	5.7
Δ % of increase		19%	87%	-16%	-46%

Figure 15. Nominal Static Safety Factors for both compressive and tensile loads

The Nominal Safety Factor of a component is always calculated in reference to the highest stress suffered. Ideally when manufacturers design a production engine, they tend to have safety factors close to the value 3.

Analysing Figure 15, it can be concluded that Tensile Loads are not critical in this engine. However, the compressive loads could compromise the production engine, and even more the modified one, in terms of fatigue performance.

Note: A table with the values of Nominal Static Safety Factor of the rod for each RPM is attached in the Appendix [Tab.1].

9 MAXIMUM COMPRESSION DISTANCE AND STRETCH IN ROD

9.1 Methodology

$$\Delta x_{\text{compression distance} | \text{stretch}} = \varepsilon * L$$

Where,

- $\Delta x_{\text{compression distance} | \text{stretch}}$ is the compression distance and stretch in the rod (mm);
- ε is the deformation of the rod cause by the Net Compressive and Tensile Loads;

$$\sigma = E * \varepsilon$$

$$\varepsilon = \frac{\sigma}{E}$$

- σ = Compression and tensile stresses (N/mm²) [Chapter 7]
 - E = Young Modulus of the rod, material's property (N/mm²);
-
- L is the rod length (mm);

9.2 Results

	Rod Length L	Powder Forged Steel E	Maximum Compression stresses	Maximum Tensile Stresses	Compressive distance Δx	Stretch Δx
Production Engine	134	170000	312	-70	0.246	-0.055
Modified Engine	134	170000	373	-131	0.294	-0.103
Δ % of increase			19%	87%	19%	87%

Figure 16. Maximum Compressive distance and Stretch in the rod

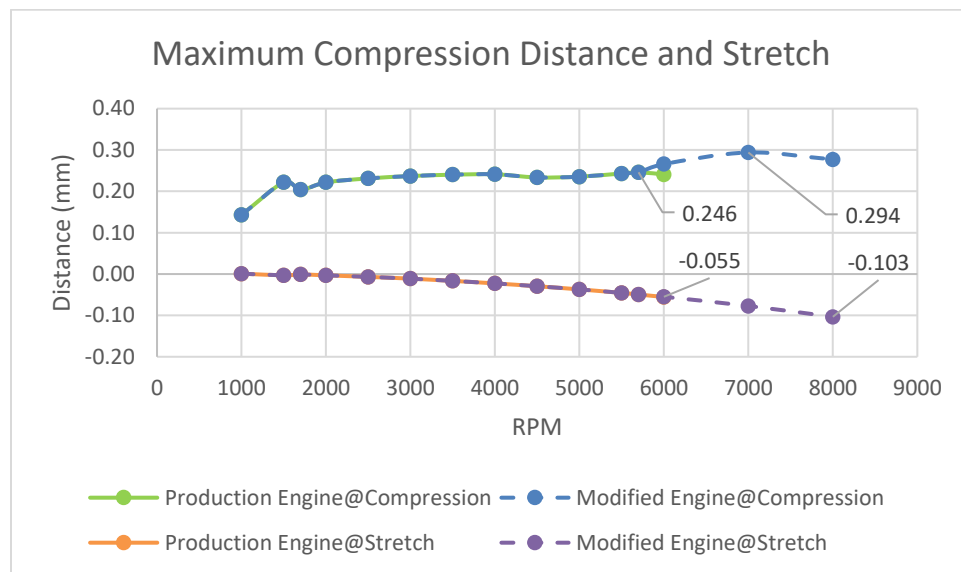


Figure 17. Maximum Compression distance and Stretch throughout the RPM range

It was proven in the previous Chapter [Chapter 8], that the conrod under maximum loads is still working in the yield region. In the yield region, metal materials present linear elastic properties, meaning that the compression distance and stretch are proportional to compression and tensile loads, respectively.

To guarantee that the engine will perform under the above conditions, the distance from the piston to the valves should be superior to the maximum stretch calculated in the conrod.

10 POWDER FORGED STEEL ROD VS TITANIUM ROD

10.1 Methodology

(Same methodology from Chapter 9 with a different value for Young Modulus)

$$\Delta x_{compression\ distance\ | \ stretch} = \varepsilon * L$$

Where,

- $\Delta x_{compression\ distance\ | \ stretch}$ is the compression distance and stretch in the rod (mm);
- ε is the deformation of the rod cause by the Net Compressive and Tensile Loads;

$$\sigma = E * \varepsilon$$

$$\varepsilon = \frac{\sigma}{E}$$

- σ = Compression and tensile stresses (N/mm²) [Chapter 7]
 - E = Young Modulus of the rod, material's property (N/mm²);
-
- L is the conrod length (mm);

10.2 Results

	Powder Forged Steel E	Powder Forged Steel UTS	Safety Factor - Compressive Loads	Safety Factor - Tensile Loads	Compressive distance Δx	Stretch Δx
Powder Forged Steel Rod	170000	750	2.0	5.7	0.294	-0.103
Titanium Rod	130000	1100	2.9	8.4	0.384	-0.135
Δ % of increase	-24%	47%	47%	47%	31%	31%

Figure 18. Modified Engine - Different conrods analysis

The results verified for Safety Factor and Compressive distance and Stretch, could be speculated solely by analysing the materials properties.

- The Titanium rod suffered a bigger stretch than the Powder Forced Steel rod due to the decrease of 24% of the Young Modulus.
- The Safety Factor from the Titanium Rod increased by the same percentage of the UTS difference between Titanium and Forged Steel.

Concluding, a Titanium rod is a good option when the maximum net loads on the rod result in stresses that could compromise the rod's performance. However, it has a drawback: the stretch observed in the Titanium rod is bigger, shortening the distance from the piston to the valves. This proximity from the piston to the valves (with the stretched rod) should be analysed to avoid a collision that will generate an engine failure.

11 CONCLUSION

The work developed allowed the author, through analytical techniques, to analyse the con-rod structural performance of a certain engine across the RPM range. A better understanding of the con-rod physics while in motion was gained, allowing relations between acceleration, forces and displacements to be made – as stated below.

A couple of aspects could be highlighted from the con-rod acceleration in relation to the overall force it suffers in a 4-stroke cycle. Firstly, the positive one, the rod acceleration helps reducing the peak vertical load during the Power Stroke, because it is generating a force acting in the opposite direction of the gas force ($F_{Inertia\ on\ rod} = M_{Piston} * acc_{conrod}$), [Chapter 4]. Secondly, between the Exhaust and Intake Stroke, where the gas forces are close to atmospheric pressure, the Inertia Forces reach their maximum values leading the con-rod to be under tensile loads, consequently deforming in that same direction.

As the RPM and cylinder pressure raises, the maximum load on the con-rod will increase, subsequently increasing the stresses in the con-rod shank leading to a possible fatigue failure. The increase in RPM will also lead to a quadratic growth of accelerations, as demonstrated in Figure 7, from Chapter 4.

This growth in acceleration will, as already stated before, generate bigger tensile loads and proportionally bigger tensile stresses [Chapter 7]. It will result in another proportional increase in stretch from the con-rod [Chapter 9] that needs to be well defined to avoid a collision between piston and valves, allowing the engine to be fully capable at its higher performance.

12 FURTHER WORK

12.1 Friction

12.1.1 Methodology

$$F_{Friction} = \mu * (F_{Side} + F_{Ring})$$

Where,

- $F_{Friction}$ is the total friction force acting inside the cylinder (N);
- μ is the friction coefficient (equal to 0.1 in cylinder walls);
- F_{Side} is the friction force caused by the piston skirt (N);

$$F_{Side} = F_{Rod} * \sin \phi$$

- F_{Ring} is the friction force caused by the piston ring (N);

$$F_{Side} = Pressure * \pi * ring\ diameter * ring\ height$$

12.1.2 Results

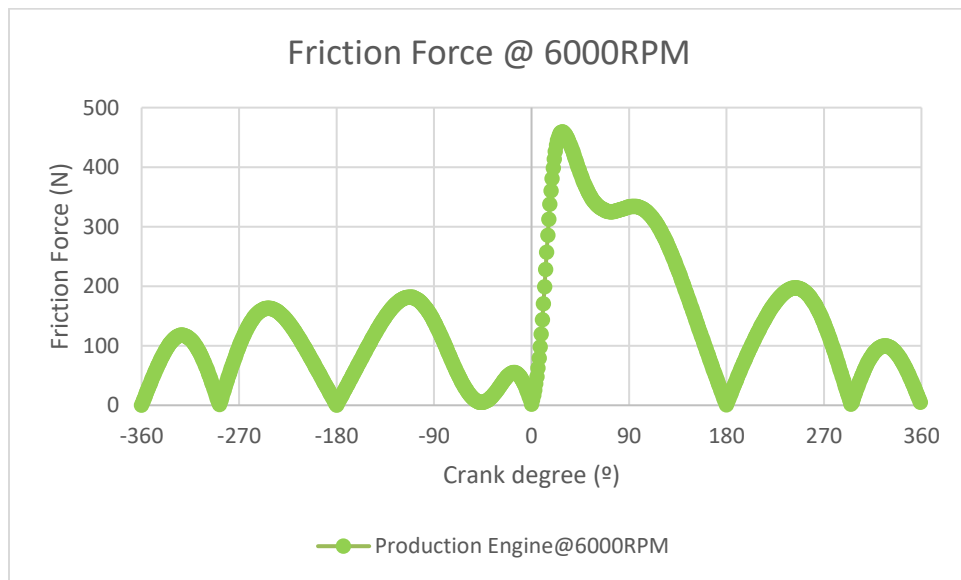


Figure 19. Example of a Friction plot for a certain RPM

Note: A table with the values of Friction Force for each RPM is attached in the Appendix [Tab.1].

12.2 Work Done

12.2.1 Methodology

$$\Delta W_{Engine} = \Delta W_{Gas} - \Delta W_{Friction}$$

Where,

- ΔW_{Engine} is the work performed by the engine during two revolutions, a 4-stroke cycle (J or Nm);
- ΔW_{Gas} is the work performed by the gas during two revolutions, a 4-stroke cycle (J or Nm);

$$\Delta W_{Gas} = \frac{F_{Gas1} + F_{Gas2}}{2} * (s_1 + s_2)$$

- $\Delta W_{Friction}$ is the work performed by friction during two revolutions, a 4-stroke cycle (J or Nm);

$$\Delta W_{Friction} = \frac{F_{Friction1} + F_{Friction2}}{2} * (s_1 + s_2)$$

12.2.2 Results

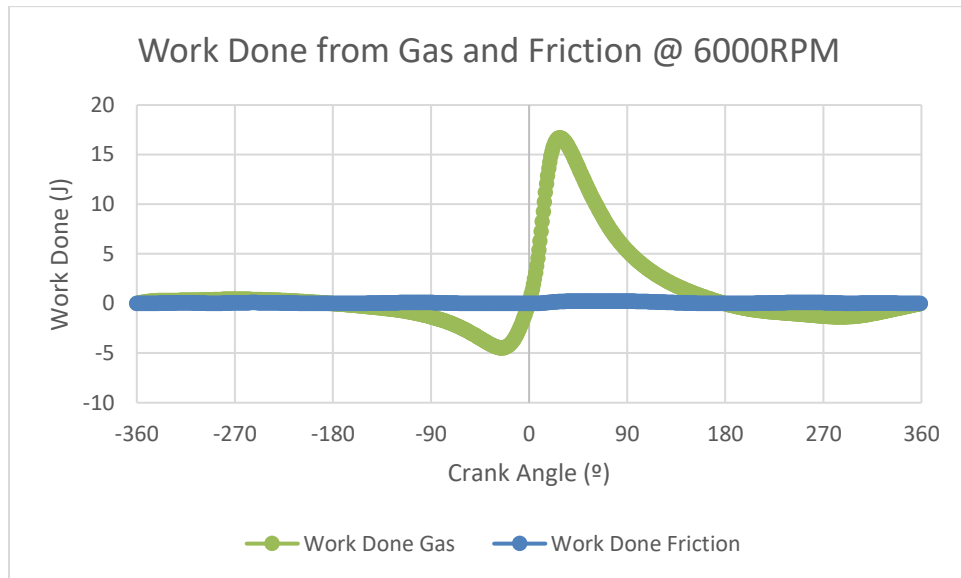


Figure 20. Work Done from Gas and Friction Force at 6000RPM

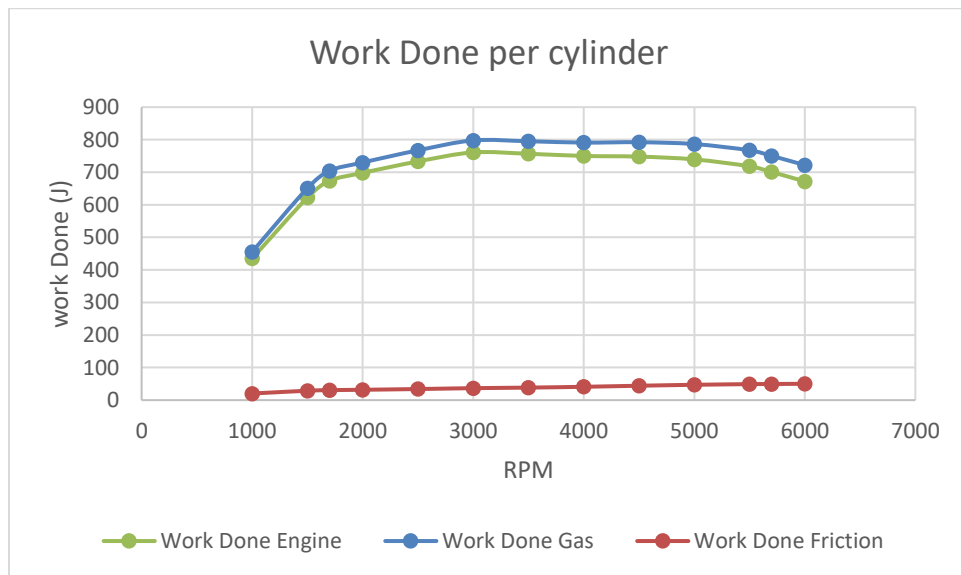


Figure 21. Work Done by cylinder throughout range of RPM

Analysing the Work Done per cylinder ($\sum |\Delta W|$) it can be concluded that the negative work done by the friction force is not substantial but increases linearly with RPM and that the profile of the Work Done by the Engine is similar to the torque curve developed by it – work done units Joules = Nm.

12.3 Mean Effective Pressure

12.3.1 Methodology

$$BMEP = NIMEP - FIMEP$$

Where,

- BMEP is Break Mean Effective Pressure;
- NIMEP is Net Indicated Mean Effective Pressure (NIMEP = GIMEP – PMEP);
- FMEP is Friction Mean Effective Pressure;

$$MEP = \frac{\sum |\Delta W|}{\text{Cylinder displacement}}$$

12.3.2 Results

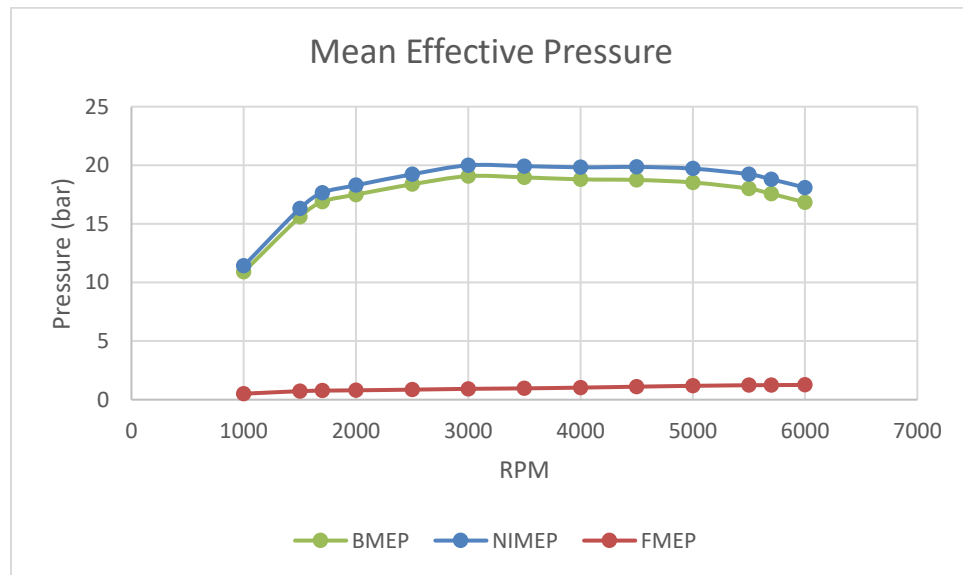


Figure 22. Mean Effective Pressure Vs RPM

For a turbo engine, the Mean Effective Pressure expected of 20 bar was achieved, proving the accuracy of the simulation developed.

Note: A table with the values of BMEP, NIMEP and FMEP for each RPM is attached in the Appendix [Tab.1].

12.4 Power

12.4.1 Methodology

$$\text{Brake Power} = \text{Net Indicated Power} - \text{Power Loss}$$

Where,

- Brake Power is the output power produced by the engine;
- Net Indicated Power is the power produced by the gas, during the Compression and Power Stroke (Gross Indicated Power) minus the Pump losses;

$$\text{Net Indicated Power} = \sum |\Delta W_{Gas}| * \frac{RPS}{2}$$

- Power Loss is the power loss from friction inside the cylinder;

$$\text{Power Loss} = \sum |\Delta W_{Friction}| * \frac{RPS}{2}$$

12.4.2 Results

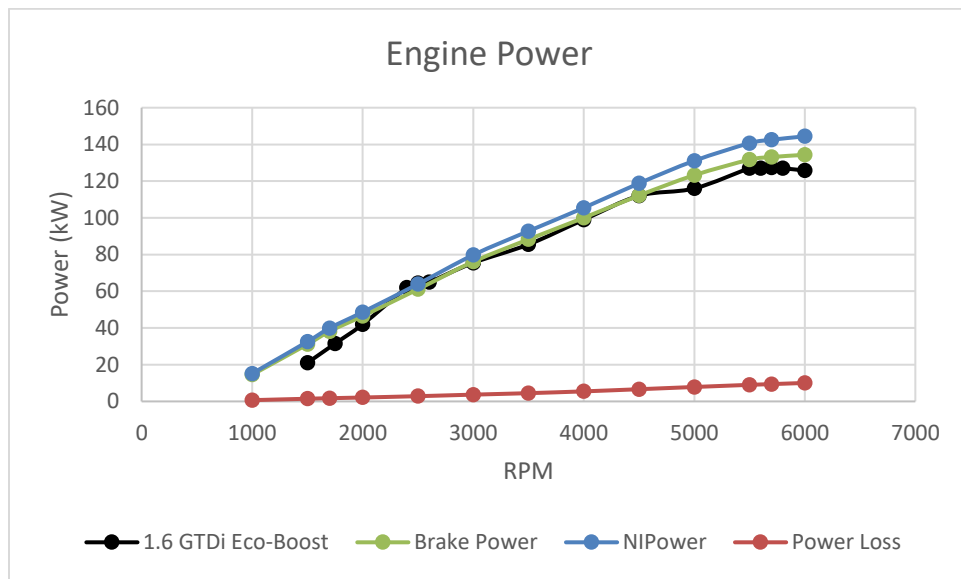


Figure 23. Engine Power Curve

The figure above, figure 23, compares the real power curve of the engine against the calculated brake power curve using the author's model. The Brake Power calculated is still slightly superior than the real power curve but that could be related to friction of some engine components not contemplated in the simulation, such as the camshaft, crankshaft, clutch, etc. The proximity of the results can confidently let the author consider his model as validated.

12.5 Torque

12.5.1 Methodology

$$\mathbf{Brake\ Torque = Brake\ Power * RPS * 2\pi}$$

Where,

Brake Torque is the torque produced by the engine from the calculated power in the previous chapter, Chapter 12.4 (Nm);

$$\mathbf{Instantaneous\ Torque = F_{Rod} * R * sin(\theta + \phi)}$$

Where,

- Instantaneous Torque is the torque produced by the force acting upon the rod per degree of crank and rod angle (Nm) – derivation attached in Appendix, Figure 28;

$$\mathbf{Gas\ Torque = F_{Gas} * R * sin(\theta + \phi)}$$

Where,

- Gas Torque is the torque produced by the gas force per degree of crank and rod angle (Nm);

$$\mathbf{Inertia\ Torque = F_{Inertia} * R * sin(\theta + \phi)}$$

Where,

- Inertia Torque is the torque produced by the inertia forces per degree of crank and rod angle (Nm);

12.5.2 Results

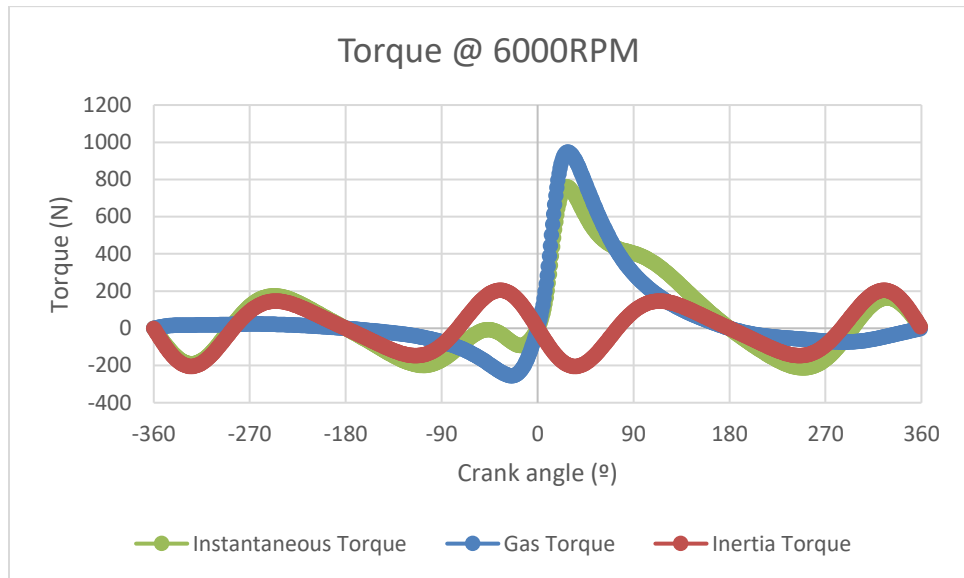


Figure 24. Torque values per degree at 6000RPM

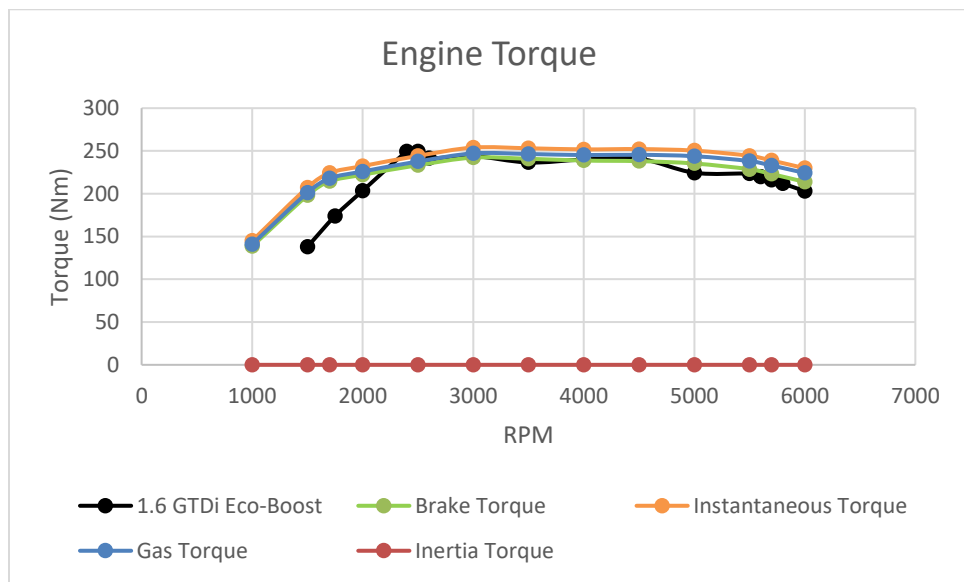


Figure 25. Engine Torque Curve

The figure above, figure 25, compares the real torque curve of the engine against the calculated brake torque curve. Like the analysis from Figure 23., Chapter 12.4.2, the brake torque curve is slightly superior to the real torque curve, due to some engine components friction not contemplated in the author's simulation. Once again, the accuracy of the results supports the validation of the simulation performed.

13 APPENDIX

	A	B	C	D	E	F	G	H	I	J	K	L
1	FROM DATA		RPM									
2	CALCULATED		6000									
3												
4	MEAN	0.00	0.00	0.00	43.85	11.21	5493.08	0.00	0.00	0.00	29.01	5493.08
5	MAX	17.68	16067.72	4880.27	81.40	95.61	46862.91	7706.77	2340.79	5512.42	5605.97	37728.26
6	MIN	-17.68	-16067.72	-4880.27	0.00	1.00	490.66	-7706.77	-2340.79	-10047.56	-10047.56	-8724.37
7	Crank Angle (°)	Conrod Angle (°)	1st order acc (m/s2)	2nd order acc (m/s2)	Piston Lift	Mean P (bar)	Gas Force (N)	1st Inertia Force	2nd Inertia Force	Inertia Forces	Inertia Forces Rod (N)	Net Force (N)
8	-360	0.00	16067.72	4880.27	0.00	2.71	1330	-7706.77	-2340.79	-10047.56	-10047.56	-8717.49
9	-359	0.30	16065.27	4877.30	0.01	2.70	1322	-7705.60	-2339.36	-10044.96	-10045.10	-8722.74
10	-358	0.61	16057.93	4868.38	0.03	2.68	1313	-7702.08	-2335.09	-10037.16	-10037.73	-8724.37
11	-357	0.91	16045.70	4853.53	0.07	2.67	1310	-7696.21	-2327.97	-10024.18	-10025.44	-8714.57
12	-356	1.21	16028.58	4832.77	0.13	2.66	1302	-7688.00	-2318.01	-10006.01	-10008.25	-8704.25
13	-355	1.52	16006.57	4806.13	0.20	2.63	1291	-7677.45	-2305.23	-9982.67	-9986.17	-8691.82
14	-354	1.82	15979.70	4773.62	0.29	2.61	1280	-7664.55	-2289.64	-9954.19	-9959.21	-8674.24
15	-353	2.12	15947.95	4735.30	0.40	2.57	1261	-7649.33	-2271.26	-9920.58	-9927.39	-8659.63
16	-352	2.42	15911.35	4691.22	0.52	2.52	1237	-7631.77	-2250.11	-9881.88	-9890.72	-8644.58
17	-351	2.72	15869.90	4641.41	0.65	2.49	1222	-7611.89	-2226.22	-9838.11	-9849.24	-8616.37
18	-350	3.02	15823.61	4585.95	0.80	2.47	1209	-7589.69	-2199.62	-9789.31	-9802.96	-8580.19
19	-349	3.32	15772.51	4524.91	0.97	2.43	1189	-7565.18	-2170.34	-9735.52	-9751.91	-8546.86
20	-348	3.62	15716.60	4458.35	1.16	2.38	1165	-7538.36	-2138.42	-9676.78	-9696.13	-8511.65
21	-347	3.92	15655.90	4386.36	1.36	2.33	1143	-7509.25	-2103.89	-9613.14	-9635.65	-8470.19
22	-346	4.21	15590.44	4309.02	1.57	2.27	1113	-7477.85	-2066.79	-9544.64	-9570.51	-8431.47
23	-345	4.51	15520.22	4226.44	1.80	2.22	1086	-7444.17	-2027.18	-9471.35	-9500.75	-8384.90
24	-344	4.80	15445.28	4138.70	2.05	2.15	1052	-7408.23	-1985.10	-9393.33	-9426.42	-8341.55
25	-343	5.09	15365.63	4045.93	2.31	2.13	1042	-7370.02	-1940.60	-9310.63	-9347.55	-8268.29
26	-342	5.39	15281.31	3948.22	2.58	2.03	995	-7329.58	-1893.74	-9223.31	-9264.21	-8228.03
27	-341	5.67	15192.32	3845.70	2.87	1.98	972	-7286.90	-1844.57	-9131.46	-9176.44	-8159.83
28	-340	5.96	15098.71	3738.50	3.18	1.94	950	-7242.00	-1793.15	-9035.15	-9084.30	-8085.45

	A	M	N	O	P	Q	R	S	T	U	V	W	X
1	FROM DATA												
2	CALCULATED												
3													
4	MEAN	5619.07	1315.22	0.04	45.28	1353.11	4.17	135.73	0.07	0.00	57.45	56.06	1.00
5	MAX	37914.08	4513.51	0.24	305.51	4560.11	35.59	459.01	0.24	204.84	761.17	948.08	16.72
6	MIN	-8724.86	0.00	-0.06	-70.31	0.00	0.37	0.07	0.00	-204.84	-212.57	-254.45	-4.48
7	Crank Angle (°)	Force Rod (N)	Side Force (N)	Stretch (mm)	Stress (N/mm2)	Side Force(N)	Force Ring (N)	Friction	Wfriction (J)	Inertia Torque/ Cyl (J) or (Nm)	Inst Torque/ Cyl (J) or (Nm)	Gas Torque/ Cyl (J) or (Nm)	Work Done by gas force
8	-360	-8717.49	0.00	-0.0554	-70.25	0.00	1.01	0.10	0.00	0.00	0.00	0.00	0.01
9	-359	-8722.86	46.24	-0.0554	-70.29	46.24	1.00	4.72	0.00	-9.30	-8.08	1.22	0.03
10	-358	-8724.86	92.48	-0.0554	-70.31	92.49	1.00	9.35	0.00	-18.58	-16.15	2.43	0.05
11	-357	-8715.67	138.54	-0.0554	-70.23	138.56	0.99	13.96	0.00	-27.83	-24.19	3.64	0.07
12	-356	-8706.20	184.46	-0.0553	-70.15	184.50	0.99	18.55	0.00	-37.01	-32.20	4.81	0.09
13	-355	-8694.87	230.17	-0.0552	-70.06	230.25	0.98	23.12	0.00	-46.11	-40.16	5.96	0.11
14	-354	-8678.62	275.53	-0.0551	-69.93	275.67	0.97	27.66	0.00	-55.12	-48.06	7.09	0.13
15	-353	-8665.57	320.76	-0.0550	-69.83	320.98	0.96	32.19	0.00	-64.01	-55.91	8.14	0.15
16	-352	-8652.31	365.74	-0.0550	-69.72	366.07	0.94	36.70	0.01	-72.76	-63.71	9.11	0.17
17	-351	-8626.11	409.86	-0.0548	-69.51	410.33	0.93	41.13	0.01	-81.36	-71.34	10.10	0.19
18	-350	-8592.15	453.17	-0.0546	-69.24	453.80	0.92	45.47	0.01	-89.78	-78.80	11.09	0.20
19	-349	-8561.25	496.16	-0.0544	-68.99	497.00	0.90	49.79	0.01	-98.02	-86.20	11.97	0.22
20	-348	-8528.67	538.58	-0.0542	-68.72	539.66	0.88	54.05	0.01	-106.05	-93.47	12.77	0.23
21	-347	-8490.03	580.08	-0.0539	-68.41	581.44	0.87	58.23	0.01	-113.85	-100.55	13.54	0.24
22	-346	-8454.33	621.22	-0.0537	-68.13	622.90	0.85	62.37	0.01	-121.42	-107.55	14.16	0.25
23	-345	-8410.93	661.20	-0.0534	-67.78	663.25	0.83	66.41	0.02	-128.73	-114.32	14.77	0.26
24	-344	-8370.94	700.81	-0.0532	-67.45	703.28	0.80	70.41	0.02	-135.78	-121.00	15.20	0.27
25	-343	-8301.08	737.16	-0.0527	-66.89	740.08	0.79	74.09	0.02	-142.53	-127.08	15.96	0.28
26	-342	-8264.51	775.69	-0.0525	-66.60	779.13	0.76	77.99	0.02	-149.00	-133.51	16.08	0.29
27	-341	-8200.02	810.86	-0.0521	-66.08	814.85	0.74	81.56	0.03	-155.15	-139.33	16.51	0.29
28	-340	-8129.43	844.50	-0.0516	-65.51	849.10	0.72	84.98	0.03	-160.99	-144.85	16.92	0.30

Figure 26. Excel sheet example with all calculations performed for each RPM

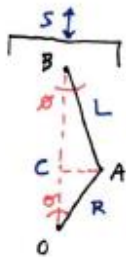
	RPM														
PEAK GAS FORCES (N)	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	7000	8000
Production engine	22527	28363	32398	35531	37575	39111	40469	41573	41324	42760	45580	46691	46863		
Modified engine	22527	28363	32398	35531	37575	39111	40469	41573	41324	42760	45580	46691	50851	58770	60178
INERTIA LOADS ON ROD (N)															
Production engine	-279	-1116	-807	-1116	-1744	-2512	-3419	-4466	-5652	-6977	-8443	-9068	-10048		
Modified engine	-279	-1116	-807	-1116	-1744	-2512	-3419	-4466	-5652	-6977	-8443	-9068	-10048	-13676	-17862
NET COMPRESSIVE LOAD (N)															
Production engine	22531	34913	32107	34913	36403	37297	37830	37966	36792	37067	38300	38746	37914		
Modified engine	22531	34913	32107	34913	36403	37297	37830	37966	36792	37067	38300	38746	41873	46280	43632
NET TENSILE LOAD (N)															
Production engine	160	-39	-170	-480	-1076	-1746	-2609	-3556	-4641	-5831	-7176	-7789	-8725		
Modified engine	160	-39	-170	-480	-1076	-1746	-2609	-3556	-4641	-5831	-7176	-7789	-8651	-12135	-16273
MAX COMPRESSION STRESSES IN ROD (N/MM2)															
Production engine	182	281	259	281	293	301	305	306	296	299	309	312	306		
Modified engine	182	281	259	281	293	301	305	306	296	299	309	312	337	373	352
MAX TENSILE STRESSES IN ROD (N/MM2)															
Production engine	1	0	-1	-4	-9	-14	-21	-29	-37	-47	-58	-63	-70		
Modified engine	1	0	-1	-4	-9	-14	-21	-29	-37	-47	-58	-63	-70	-98	-131
STATIC SAFETY FACTOR FOR COMPRESSION LOADS															
Production engine	4.13	2.67	2.90	2.67	2.56	2.50	2.46	2.45	2.53	2.51	2.43	2.40	2.45		
Modified engine	4.13	2.67	2.90	2.67	2.56	2.50	2.46	2.45	2.53	2.51	2.43	2.40	2.22	2.01	2.13
STATIC SAFETY FACTOR FOR TENSILE LOADS															
Production engine	0.0	2389.1	547.2	194.1	86.5	53.3	35.7	26.2	20.1	16.0	13.0	11.9	10.7		
Modified engine	0.0	2389.1	547.2	194.1	86.5	53.3	35.7	26.2	20.1	16.0	13.0	11.9	10.8	7.7	5.7
MAX COMPRESSION DISTANCE (MM)															
Production engine	0.143	0.222	0.204	0.222	0.231	0.237	0.240	0.241	0.234	0.235	0.243	0.246	0.241		
Modified engine	0.143	0.222	0.204	0.222	0.231	0.237	0.240	0.241	0.234	0.235	0.243	0.246	0.266	0.294	0.277
MAX STRETCH (MM)															
Production engine	0.001	-0.003	-0.001	-0.003	-0.007	-0.011	-0.017	-0.023	-0.029	-0.037	-0.046	-0.049	-0.055		
Modified engine	0.001	-0.003	-0.001	-0.003	-0.007	-0.011	-0.017	-0.023	-0.029	-0.037	-0.046	-0.049	-0.055	-0.077	-0.103
TITANIUM MAX COMPRESSION DISTANCE (MM)															
Production engine	0.187	0.290	0.267	0.290	0.302	0.310	0.314	0.315	0.306	0.308	0.318	0.322	0.315		
Modified engine	0.187	0.290	0.267	0.290	0.302	0.310	0.314	0.315	0.306	0.308	0.318	0.322	0.348	0.384	0.362
TITANIUM MAX STRETCH (MM)															
Production engine	0.001	0.000	-0.001	-0.004	-0.009	-0.015	-0.022	-0.030	-0.039	-0.048	-0.060	-0.065	-0.072		
Modified engine	0.001	0.000	-0.001	-0.004	-0.009	-0.015	-0.022	-0.030	-0.039	-0.048	-0.060	-0.065	-0.072	-0.101	-0.135
TITANIUM - STATIC SAFETY FACTOR FOR COMPRESSION LOADS															
Production engine	6.06	3.91	4.25	3.91	3.75	3.66	3.61	3.60	3.71	3.68	3.56	3.52	3.60		
Modified engine	6.06	3.91	4.25	3.91	3.75	3.66	3.61	3.60	3.71	3.68	3.56	3.52	3.26	2.95	3.13
TITANIUM - STATIC SAFETY FACTOR FOR TENSILE LOADS															
Production engine	0.0	3504.0	802.6	284.6	126.9	78.2	52.3	38.4	29.4	23.4	19.0	17.5	15.6		
Modified engine	0.0	3504.0	802.6	284.6	126.9	78.2	52.3	38.4	29.4	23.4	19.0	17.5	15.8	11.2	8.4

Table 1. Maximum Values per RPM from calculations performed in the report

	RPM													
FRICTION	1000	1500	1700	2000	2500	3000	3500	4000	4500	5000	5500	5700	6000	
Work Done by friction (J)	20.31	28.90	30.75	31.80	34.09	36.80	38.47	40.97	44.22	47.12	49.14	49.57	50.25	
Power Loss/cyl (kW)	0.17	0.36	0.44	0.53	0.71	0.92	1.12	1.37	1.66	1.96	2.25	2.35	2.51	
Power Loss (kW)	0.68	1.44	1.74	2.12	2.84	3.68	4.49	5.46	6.63	7.85	9.01	9.42	10.05	
GAS														
Work Done by Gas (J)	455.49	650.98	704.37	729.85	767.23	797.72	795.12	790.99	792.17	786.52	767.34	750.25	721.95	
Indicated Power/cyl (kW)	3.80	8.14	9.98	12.16	15.98	19.94	23.19	26.37	29.71	32.77	35.17	35.64	36.10	
Indicated Power (kW)	15.18	32.55	39.91	48.66	63.94	79.77	92.76	105.47	118.82	131.09	140.68	142.55	144.39	
BRAKE POWER														
Work Done engine (J)	435.18	622.08	673.61	698.05	733.14	760.92	756.66	750.02	747.95	739.40	718.20	700.67	671.71	
Brake Power (kW)	14.51	31.10	38.17	46.54	61.10	76.09	88.28	100.00	112.19	123.23	131.67	133.13	134.34	
BRAKE TORQUE														
Torque (Nm)	138.52	198.02	214.42	222.20	233.37	242.21	240.85	238.74	238.08	235.36	228.61	223.03	213.81	
Gas Torque (Nm)	140.95	201.20	217.91	226.00	237.65	247.13	246.43	245.22	245.50	243.86	238.14	232.94	224.26	
Instantaneous Torque (Nm)	145.00	207.23	224.22	232.33	244.23	253.94	253.11	251.80	252.17	250.37	244.27	238.83	229.82	
Inertia Torque (Nm)	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	
MEP														
FMEP (bar)	0.51	0.72	0.77	0.80	0.85	0.92	0.96	1.03	1.11	1.18	1.23	1.24	1.26	
NIMEP (bar)	11.42	16.32	17.65	18.29	19.23	19.99	19.93	19.82	19.85	19.71	19.23	18.80	18.09	
BMEP (bar)	10.91	15.59	16.88	17.50	18.37	19.07	18.96	18.80	18.75	18.53	18.00	17.56	16.83	

Table 2. Maximum values per RPM from calculations done in Chapter 12. Further Work

PISTON ACCELERATION



$$S = R + L - OB$$

$$OB = OC + CB$$

$$\triangle OAC ; OC = R \cos \theta$$

$$\triangle ABC ; CB = L \cos \phi$$

$$OB = R \cos \theta + L \cos \phi$$

$$S = R + L - R \cos \theta - L \cos \phi$$

$$\triangle OAC ; AC = R \sin \theta$$

$$\triangle ABC ; AC = L \sin \phi$$

$$R \sin \theta = L \sin \phi \Leftrightarrow \sin \phi = \frac{R}{L} \sin \theta ; \text{ because } \cos^2 \phi + \sin^2 \phi = 1$$

$$\cos \phi = \left(1 - \frac{R^2}{L^2} \sin^2 \theta \right)^{1/2}$$

$$S = R + L - R \cos \theta - L \left(1 - \frac{R^2}{L^2} \sin^2 \theta \right)^{1/2}$$

↓ APPROXIMATION

$$S = R + L - R \cos \theta - L \left(1 - \frac{1}{2} \frac{R^2}{L^2} \sin^2 \theta \right)$$

↓ ACCELERATION OF PISTON

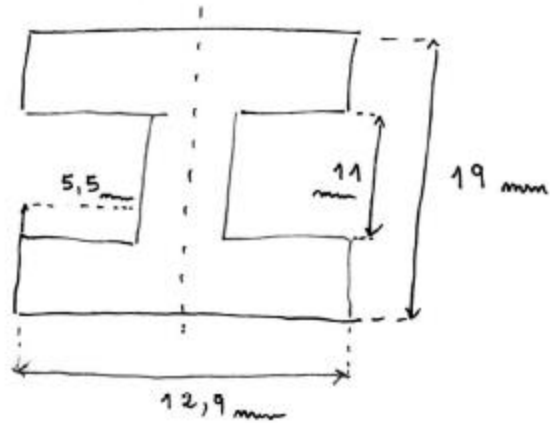
$$\frac{d^2 S}{d\theta^2} = R \cos \theta + \frac{R}{L} \cdot R \cos 2\theta + \text{SMART TERMS}$$

↓ BECAUSE $\frac{ds}{dt} = \frac{ds}{d\theta} \times \omega ; \omega = 2\pi \frac{\text{RPM}}{60}$

$$\frac{d^2 S}{dt^2} = R \omega^2 \cos \theta + \frac{R}{L} \cdot R \omega^2 \cos 2\theta$$

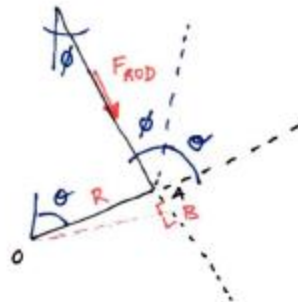
Figure 27. Conrod Angle and Piston Acceleration Derivation

CON-ROD SHANK CROSS-SECTIONAL AREA



$$\text{AREA} = (12,9 \times 19) - 2(5,5 \times 11) = 124,1 \text{ mm}^2$$

INSTANTANEOUS TORQUE



$$OA = R ; \angle OAB = \theta + \phi$$

$$\triangle OAB$$

$$OB = OA \cdot \sin(\theta + \phi)$$

$$OB = R \cdot \sin(\theta + \phi)$$

$$\text{INST TORQUE} =$$

$$= F_{\text{ROD}} \cdot R \cdot \sin(\theta + \phi)$$

Figure 28. Rod shank cross-sectional area and Instantaneous Torque Derivation

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